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A BASIC INVESTIGATION OF THE

FULL FLOATING TEXTILE SPINDLE BEARING

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A THESIS

Presented to

the Faculty of the Graduate Division

by

Richard Dean Cheverton

In Partial Fulfillment

of the Requirement for the Degree

Master of Science in Mechanical Engineering

Georgia Institute of Technology

June 1953

A BASIC INVESTIGATION OF THE

FULL FLOATING TEXTILE SPINDLE BEARING

Approved:

Date Approved by Chairman: June 3, 1953

ACKNOWLEDGMENTS

On the completion of this investigation I wish to express my sincere thanks to the following: To Dr. R. L. Sweigert, Dean of the Graduate Division; to The Texas Company and Mr. Harry R. Robinson, Assistant Director of Lubricating Oil Research Department; to Dr. J. P. Vidosic, my thesis advisor; to Dr. J. L. Taylor, Professor of Textile Engineering; and to Professor H. A. Dickert, Director of the Textile Engineering School, for making possible and for aiding me in the pursuance of this research.

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SUMMARY

The purpose of this investigation was to determine the region of lubrication within which the bearings in a plain cast-iron bolster type cotton spinning spindle operate so as to facilitate the choice of a proper lubricant for optimum operating conditions.

In accomplishing the above it was necessary to develop the apparatus capable of accurately measuring the magnitudes of the variables involved in the theory of hydrodynamic lubrication. An attempt was first made to adapt a dynamometer previously developed for research pertaining to the measurement of the power consumption of a cotton spinning spindle. The dynamometer proved inadequate because it had to measure many friction losses besides the bearing losses. All significant figures were lost in the attempt to separate the bearing friction from the total friction measured.

The present friction torque measuring device is capable of measuring only the friction in the bearings of the spindle. This was accomplished by mounting the spindle bearings on a "frictionless" mount so that the friction force applied to the spindle bearings was measured as a restraining force on the outside of the bearings. This restraining force was applied by means of a small cantilever beam equipped with two SR-4 electrical strain gages and a thread which was wrap-

ped around the fixture holding the spindle bearings. A continuous recording was made with a strain time recording instrument. Once this piece of apparatus was calibrated, the strain reading on the recording chart was directly proporional to both the friction torque and friction force on the spindle bearings. Since the friction on the journal, rather than the bearing, is desirable a simple relationship involving the eccentricity of the journal with respect to the bearing was used to convert the measured value to journal friction.

The plain cast-iron bolster type of spindle contains two bearings. One is a plain journal type while the other is a conical pivot type, called the step bearing. The friction on these two cannot be measured separately. After considering the type of loading on the spindle, the assumption was made that the step bearing consumed a relatively small amount of the spindle power. This assumption proved to be justifiable.

The bearings were analyzed by comparing their performance with the hydrodynamic theory of lubrication and published experimental data for the two types of bearings. A theoretical solution of the step bearing does not exist. Considering the two bearings as one unit, use was made of the Sommerfeld curves to determine if the unit was operating in the hydrodynamic or boundary region of lubrication. This was sufficient to accomplish the purpose of the investigation.

The results obtained indicated that the journal bearing was operating hydrodynamically, while the step bearing was operating in the region of extreme boundary lubrication. Under these operating conditions a boundary type lubricant would be necessary to decrease the wear and power consumption of the step bearing.

Since during this investigation the spindle was not subjected to all of its normal operating loads, it is recommended that the work be extended to include all of the generally encountered operating conditions. In addition it is suggested that consideration be given to the theory of oil whip.

In this way only, may the proper selection of a lubricant be made.

CHAPTER I

INTRODUCTION

In any plant where large machines are used a relatively large amount of power is expended for no useful purpose. This power is used to overcome the friction in bearings, the energy being dissipated as heat.

It is seldom obvious to the owners of these machines that such a large portion of their overhead is due to bearing friction. For this reason a great deal of work has been done toward educating those directly responsible for the maintenance of the machines. They are generally told which lubricant is best suited for a particular bearing. In many cases this procedure is adequate because a bearing that has been properly designed and is continuously subjected to the predicted load needs only to be properly maintained to insure optimum operating characteristics. However, there are an unestimable number of bearings in service today that, for various reasons, are not operating at their optimum. The reasons may be that the bearing was originally improperly designed, or that the load or speed is other than predicted. To overcome faults of this type the bearing needs to be redesigned. Much too often this is accomplished by the production of a newer model machine; yet the old machines are still in operation, performing under the up-to-date speeds and loads.

Taking the preceding facts into consideration, it appears justifiable to investigate the bearings in overburdenned machinery in an attempt to reduce wasteful power consumption as well as to improve general performance.

One of the largest industries today that has found it necessary to continue to operate their old machinery is the textile industry. It is true that recently several creditable improvements have been incorporated into the latest textile equipment. A certain amount of the improvement has been on the bearings which exist by the thousands in textile machines. Nevertheless, due to the abnormal competitive nature of the industry and the never ceasing demands for their products, much of the original machinery remains in constant use. It seems logical, therefore, that a great deal of power is being used needlessly to overcome friction in bearings that were not designed for their present operating conditions.

A piece of ancient textile equipment is found in the spinning room where a larger portion of the plant's power is consumed. It is the cotton spinning spindle, which is located on the spinning frame. Since there are usually several thousand spindles operating in one spinning room it appears logical to investigate the bearings in the spindle.

The spindles generally encountered in cotton mills are of the following types:

1. Spindles with plain cast-iron bolsters, and cast-iron steps and guides;

2. Spindles with plain steel tubing bolsters, and with porous-metal inserts for steps and guides;

3. Spindles with plain steps and ball- or roller-bearing guides;

4. Spindles equipped entirely with ball-bearings.

Many of the new frames are equipped with the ball-bearing spindles. However, thousands of the first two types of spindles are in operation today and are being sold as replacements. At this time the plain bolster type of spindle is predominant.

When attempting to investigate a lubricated bearing, the basic theory of lubrication must be considered. The so-called "perfectly" lubricated bearing is one that functions according to the hydrodynamic theory of lubrication. Bearings that do not function thusly are considered to operate with boundary lubrication. The types of lubricants to be used in either case can be radically different.

It is quite possible that the proper type of lubricant could not be picked at random. In the case of a hydrodynamic bearing the viscosity of the lubricant is generally the property to be considered. The bearing and journal are completely separated by a fluid film, the source of friction being the viscous drag imposed by shear within the lubricant. However, if the bearing is operating in the boundary region of lubrication the surface finish and the physical and chemical properties of the surfaces and the lubricant become important. Boundary lubricants are generally produced by the addition of a relatively small quantity of a polar organic chemical to a hydrocarbon lubricating oil.

In the past investigators have obtained power curves for plain bolster spindles, but this information alone is not sufficient for determining the true running conditions of the bearings. Four graduate students at the Georgia Institute of Technology[1,2,3,4] have participated in research work connected with the power consumption of the plain bolster spindle. A check of this and other published data shows considerable disagreement. Actually, this is understandable because there are so many variables that affect the performance of a lubricated bearing. It would be difficult to accurately reproduce the same results, even when using the same test-apparatus. Jones[5] has found that vibration or wobble of a spindle very decidedly affects the power consumption of a spindle. The magnitude of this vibration is very difficult to control for test purposes.

The plain bolster contains two bearings, as indicated before. The step or thrust bearing is of the conical type. The other is a plain journal bearing. According to Jones, the power consumed by the step bearing increases as additional weight is added to the spindle or package. There apparently is no data giving the actual magnitude of the power consumed by the step bearing or the journal bearing alone. There appears to be no published theory for a lubricated conical pivot bearing, and also no data exists showing the relationship between power and belt tension, which would be an indication of the rate of change of power consumption in the journal bearing.

Performance of bearings.—As mentioned previously a bearing operates either in the boundary or hydrodynamic region of lubrication. This is diagramatically illustrated by the curve in figure 1. This curve is essentially a plot of the coefficient of friction against the Sommerfeld number[6]. The Sommerfeld number is a dimensionless combination of the variables r, c, u, N and P that appears in the theory of hydrodynamic lubrication in the following form:

$$\frac{r^2}{c^2} \cdot \frac{uN}{P}$$

where r = radius of journal

c = radial clearance between journal and bearing

u = absolute viscosity

N = speed, rps

P = Load per unit projected bearing area.

The coefficient of friction is defined by the expression:

$$f = \frac{F_j}{W}$$

where F_j = friction force on the journal

W = total load on the bearing.

To establish the type of lubrication a bearing is operating with the Sommerfeld number and friction force should be determined experimentally, using them to plot a curve simi-

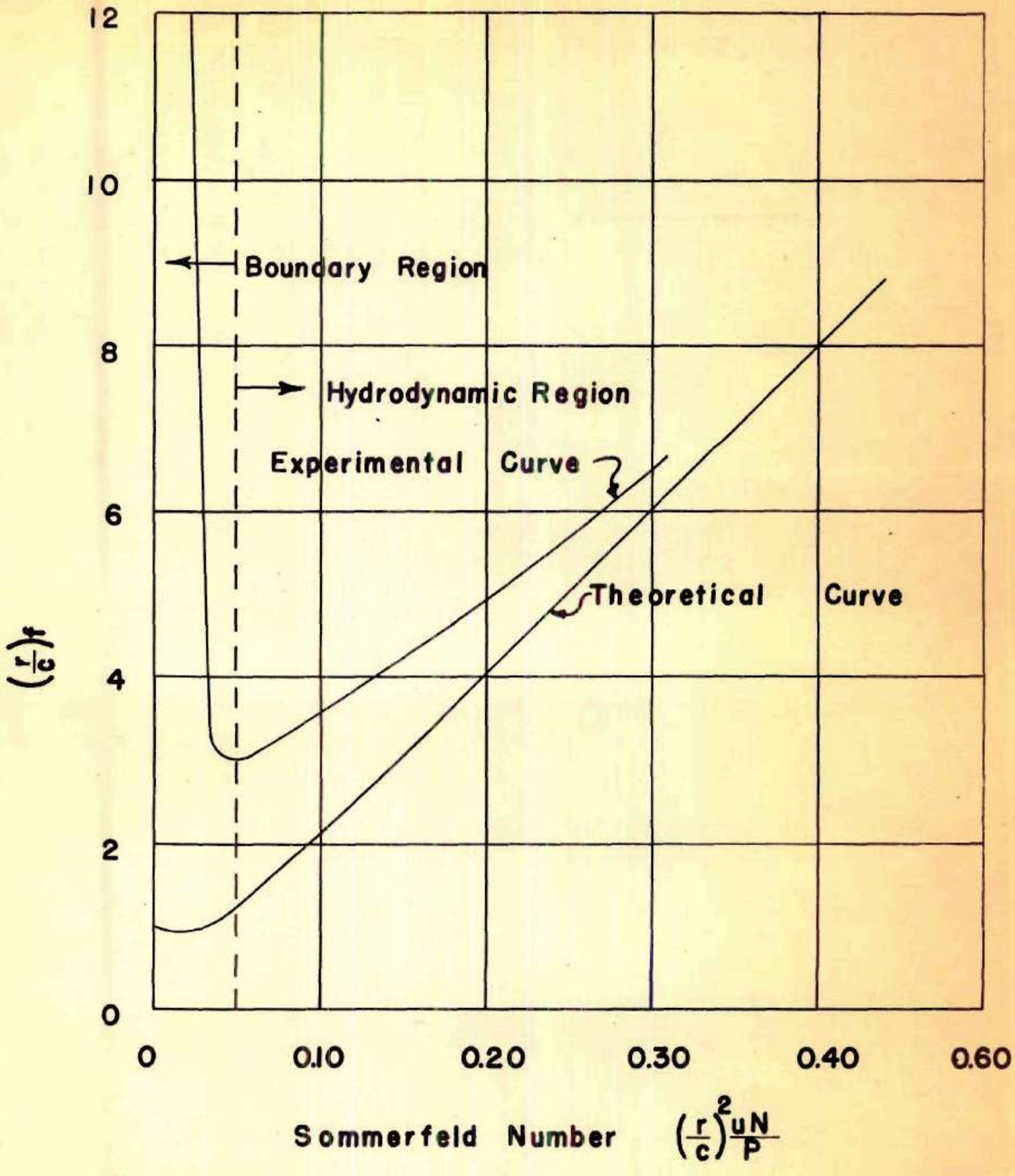


Fig. 1. Friction Curves
Full Journal Bearing

lar to the one in figure 1. The type of lubrication present should then be evident.

Oftentimes in this type of experimental work the friction force on the bearing rather than on the journal is measured. One may be converted to the other by the following relationship:

$$\frac{F_b}{F_j} = \frac{1-n^2}{1+2n^2}$$

This expression may be obtained from the two equations involving the friction forces on the bearing and journal. They are:

$$\frac{r}{c}\frac{uN}{F'_j} = \frac{(2+n^2)(1-n^2)^{\frac{1}{2}}}{4\pi^2(1+2n^2)} \quad ; \quad \frac{r}{c}\frac{uN}{F'_b} = \frac{(2+n^2)}{4\pi^2(1-n^2)^{\frac{1}{2}}}$$

where F'_j = friction force on journal per unit projected bearing area

F'_b = friction force on bearing per unit projected bearing area

n = attitude of the journal.

Values of n can be obtained for a full journal bearing from the following equation, by calculating values of the Sommerfeld number for various values of n and plotting an s-n curve.

$$S = \frac{(2+n^2)(1-n^2)^{\frac{1}{2}}}{12\pi^2 n}$$

Limitations on obtaining necessary data.—The design of the plain bolster prohibits the measurement of the friction forces of the two separate bearings. The friction measured is the total friction force for both bearings. If the bearings appear to be operating hydrodynamically, an approximation of the individual friction forces may be obtained by using theoretical equations to calculate the friction in the journal bearing.

Published experimental data[7] on conical thrust bearings indicates that the step bearing can operate hydrodynamically. Therefore, the preceding suggested method seems feasible.

The absolute viscosity of the lubricant, as it appears in the theoretical equations, is the average viscosity of the oil film in the bearing. To determine this viscosity it is necessary to determine the temperature of the oil film. A method for doing this is suggested by Muskat and Morgan[8]. They mounted thermocouples in the surface of the bearing in question and measured the temperature directly. Due to the relatively small size of the bearings in the spindle it would be very difficult and possibly detrimental to the operation of the bearings to attempt to imbed thermocouples in the bearing surface. However, it seems feasible to measure the temperature on the back side of the bearing, assume the heat dissipated through the bearing, and use heat transfer theory to establish an approximate oil film temperature.

When in actual service, the bearings in the plain bolster are subjected to complicated and varying loads. The step bearing carries not only the thrust load, but also a radial load due to the tension in the thread. The journal bearing is subjected to the belt tension and thread tension load. Also a certain amount of wobble or vibration of the spindle takes place due to the unbalance of the bobbin. Jones[5] shows that the power consumption is increased considerably with an increase in amplitude of spindle vibration. For this particular investigation, however, an attempt will be made to keep the amplitude of vibration at a minimum. Also no thread tension load will be applied to the spindle. These conditions will be left for further investigation.

Purpose of investigation.—The purpose of the investigation is as follows:

1. To develop the apparatus which is capable of measuring the variables involved in the hydrodynamic theory of lubrication;

2. To obtain the necessary data for attempting an analysis of the plain bolster type bearings;

3. To compare the experimental results with the hydrodynamic theory;

4. To attempt to establish the region of lubrication in which the plain bolster spindle operates.

It is hoped that in the future the investigation will be extended to include the thread tension load and vibration effects. In this way it may be possible to determine an optimum range of operation, and an appropriate type lubricant, or possibly a design modification.

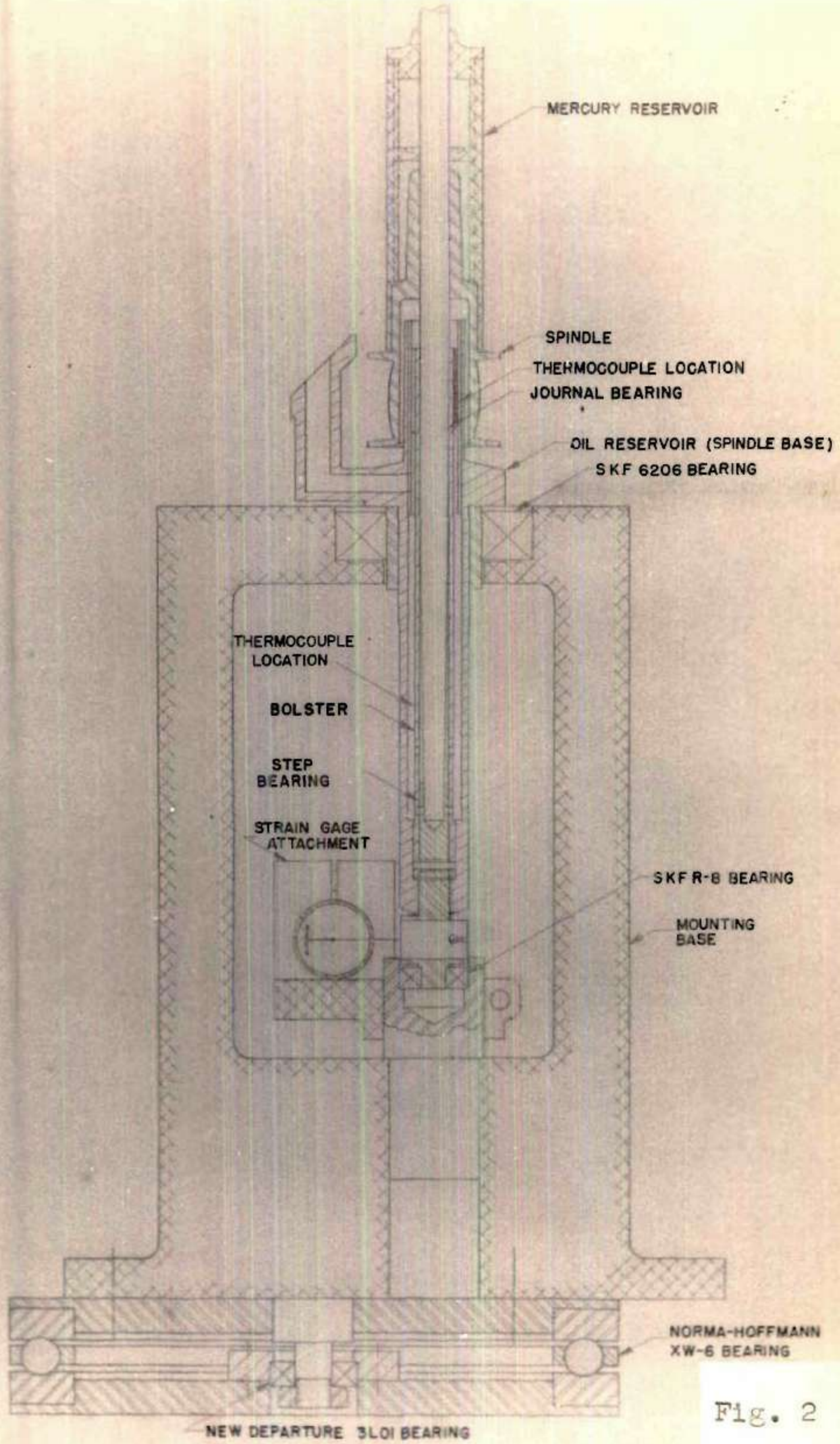
CHAPTER II

INSTRUMENTATION AND EQUIPMENT

In 1949 Robert L. Newell[2], a graduate student at the Georgia Institute of Technology, with the aid of technical personnel at the West Point Manufacturing Company developed a dynamometer for the purpose of measuring the power consumption of a cotton spinning spindle. The dynamometer is capable of measuring the total input power, which includes the belt windage losses, the idler pulley bearing and windage losses, and the bearing and windage losses of the spindle. This dynamometer was used by three of the Georgia Institute of Technology graduate students[2,3,4] referred to previously.

An attempt was made to adapt the same apparatus for this investigation. It was found, however, that after separating the bearing power from the total input power all significant figures were lost.

It then became necessary to design a torque measuring device that would measure only the friction torque of the bearings in the bolster. This was accomplished by mounting the oil reservoir on two ball-bearings as shown in figure 2. The turning moment of the reservoir due to the friction force on the spindle bearings was restricted by a thread wrapped about an extension at the base of the oil reservoir. The other end of this thread was fastened to the free end of a



SPINDLE DYNAMOMETER

Fig. 2

Figure 3

Spindle Dynamometer - Assembly

1. Mounting Base
2. Mercury Reservoir
3. Thermocouple Leads - Oil Reservoir
4. Thermocouple Leads - Journal Bearing
5. Thermocouple Connecting Post
6. Chord for Maintaining Belt Tension
7. Friction Torque Measuring Device
8. Pulley used in Calibrating Friction Torque Measuring Device
9. Thrust Bearing Assembly

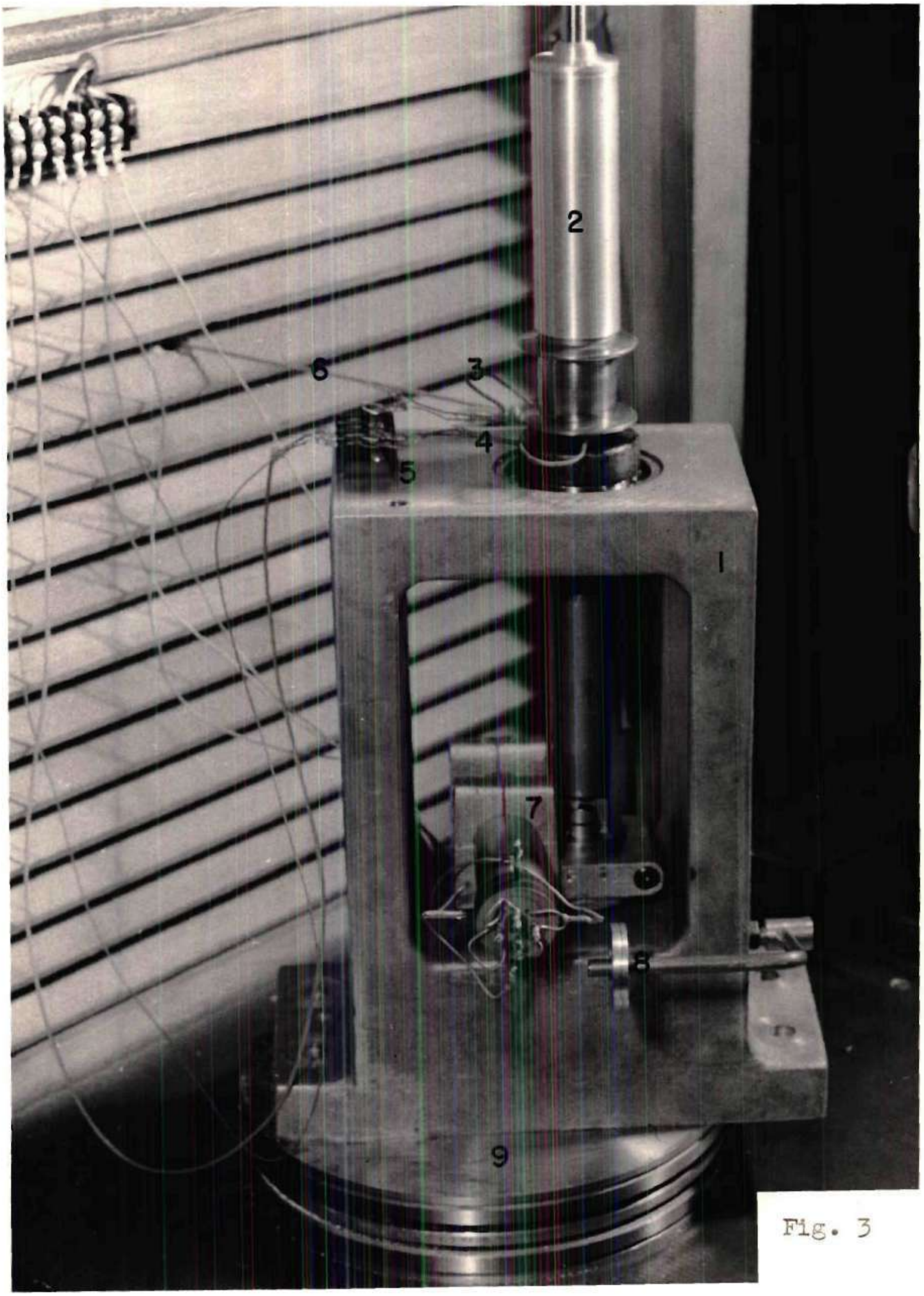


Fig. 3

Figure 4

Spindle Dynamometer - Operating Equipment

1. Dynamometer
2. Chord for maintaining belt tension
3. Spindle Drive Belt
4. Idler Pulley
5. Stroboscope
6. Nine Speed Transmission
7. Cast-iron Rubber Mounted Base
8. Point of Journal Load Application
9. Thrust Bearing Adjustment

Fig. 4

Figure 5

Spindle Dynamometer - Operating Equipment

1. Voltage Regulator
2. Ammeter in Driving Motor Circuit
3. By-pass Switch to Ammeter
4. Point of Journal Load Application
5. Synchronous Motor
6. Mechanism for Adjusting Thrust Bearing

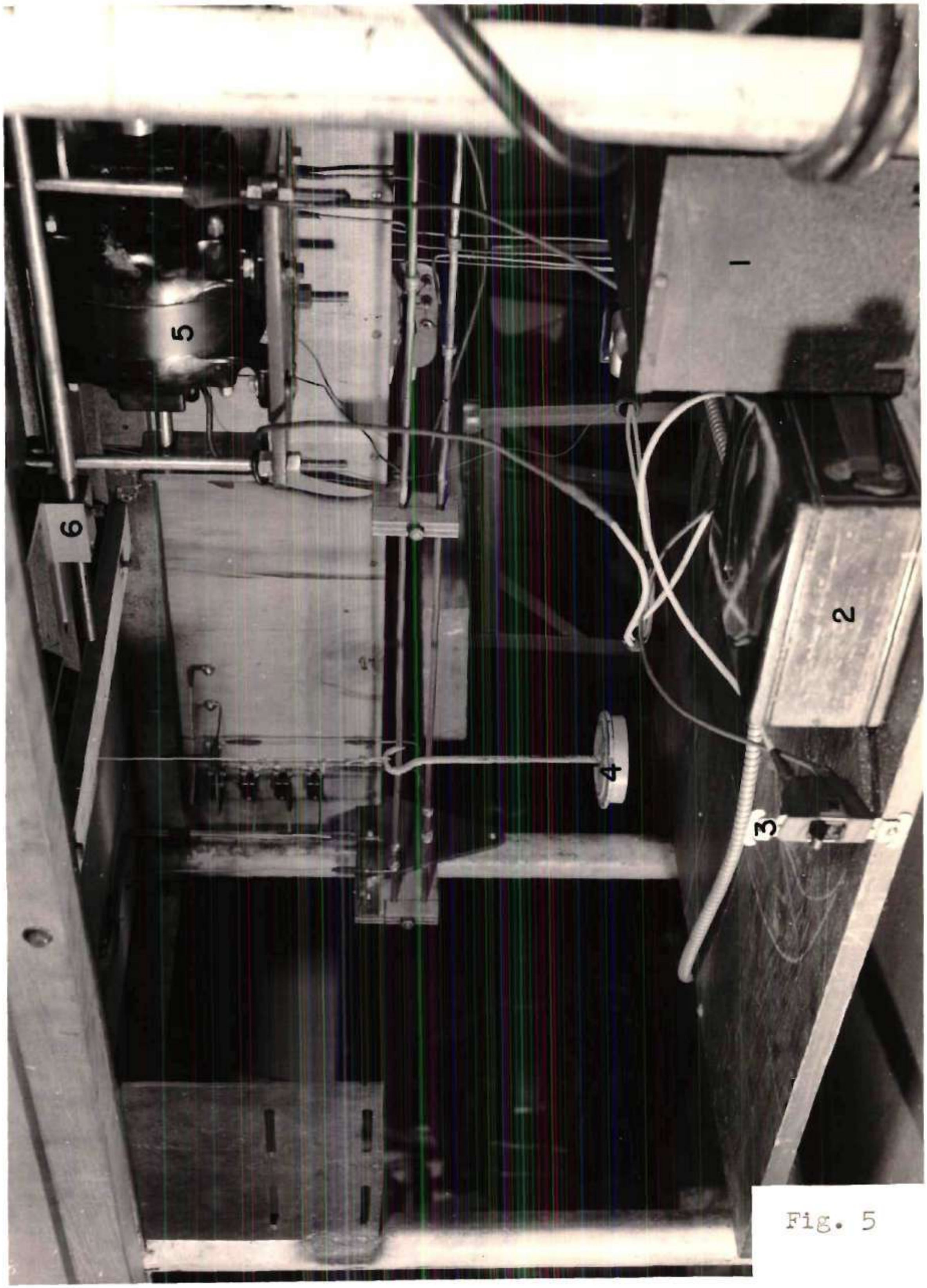


Fig. 5

Figure 6

Spindle Dynamometer - Component parts of Spindle and Friction

Torque Measuring Device

1. Cantilever Housing
2. Strain Measuring Element
3. Spindle and Mercury Reservoir
4. Cast-Iron Bolster with Journal Bearing Thermocouple
5. Special Spindle Oil Reservoir

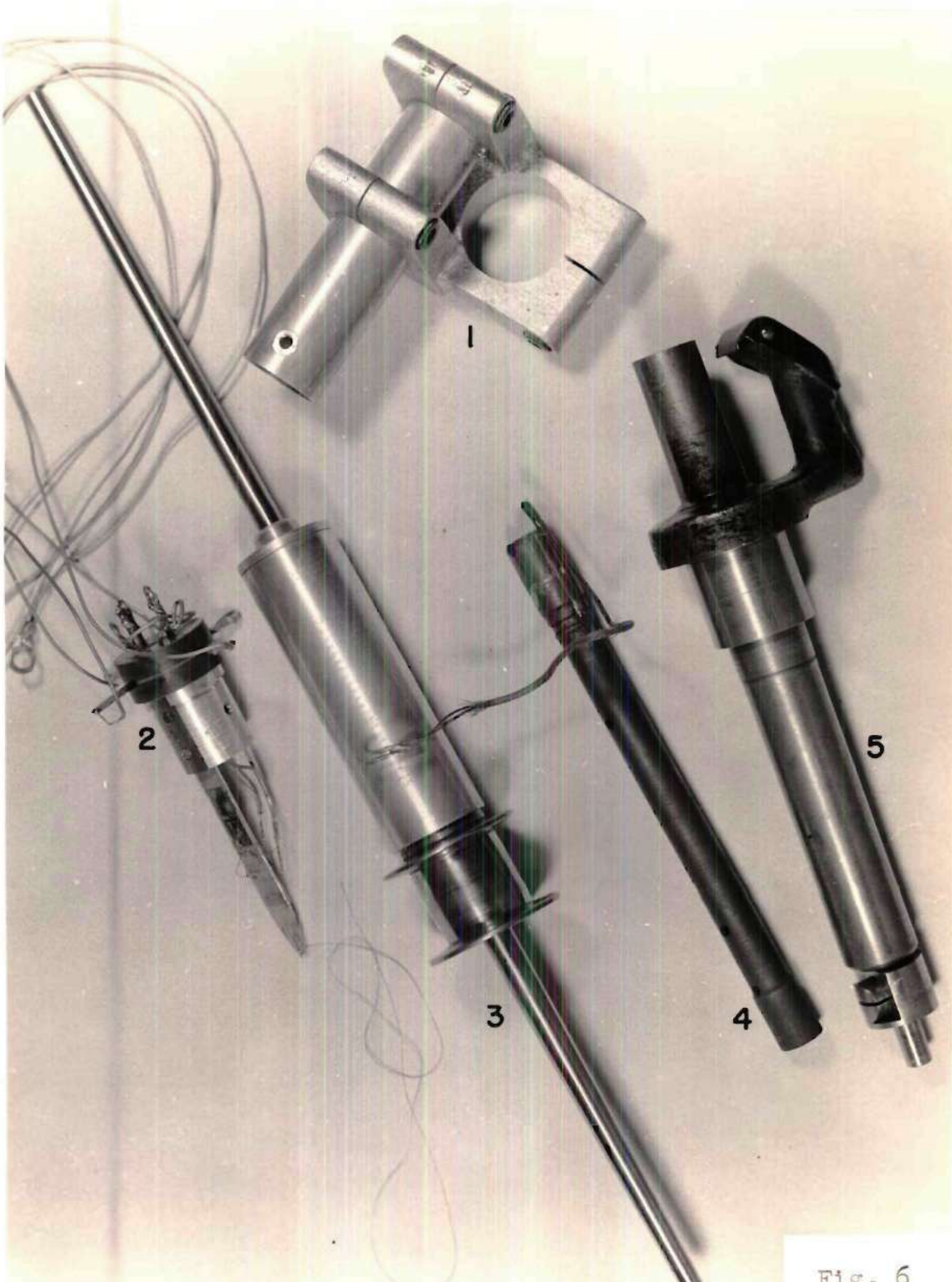


Fig. 6

Figure 7

Spindle Dynamometer - Controls and Recording Instruments

1. Foxboro SR-4 Strain-Time Recorder
2. Foxboro portable Indicator (Potentiometer used with Thermocouples)
3. Rheostat and Switches for Controlling Air Conditioner
4. Stroboscope
5. Heating Elements and Fan Housing
6. Regulated Voltage Switch and Outlet
7. Thermocouple Selector Switch

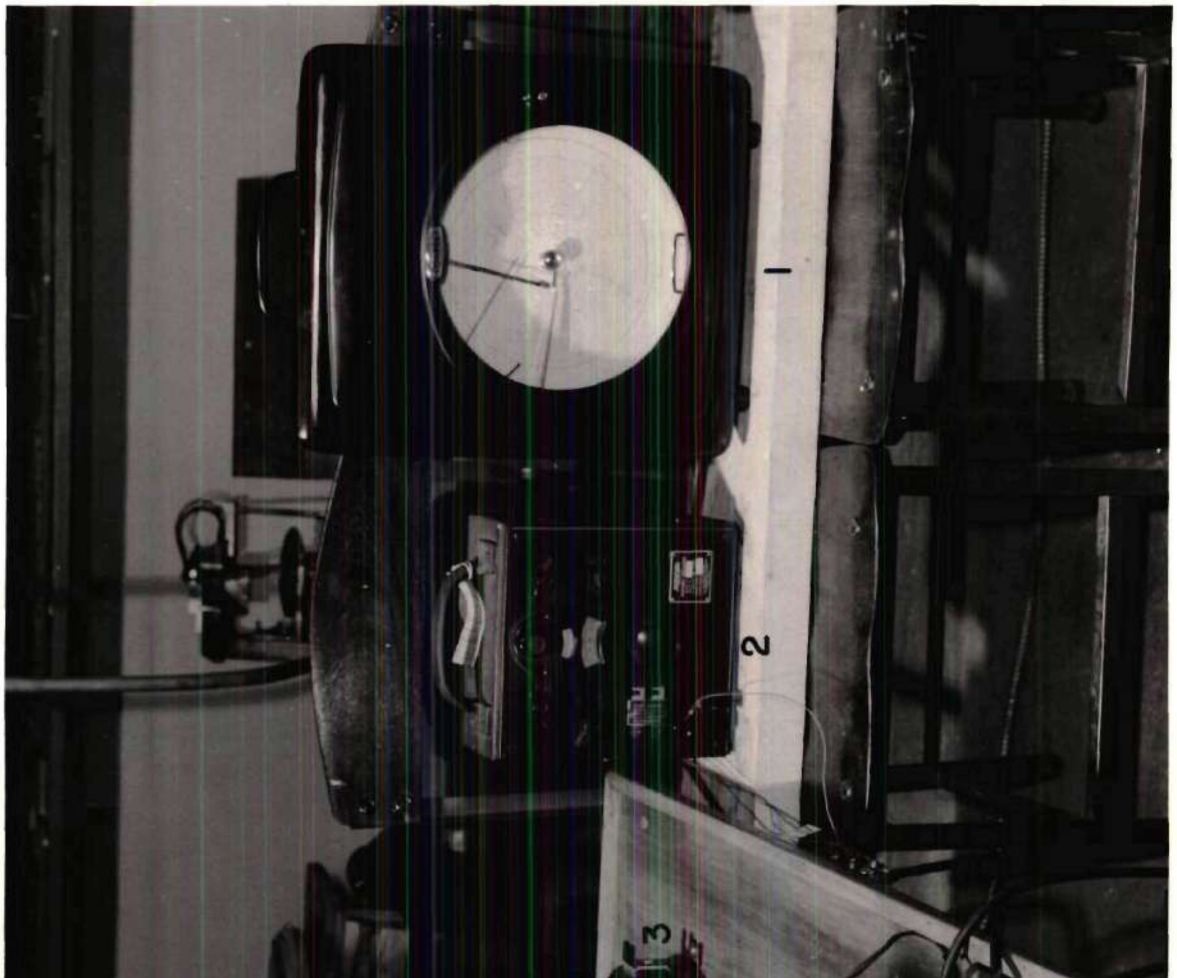


Fig. 7

small cantilever beam that had two SR-4 electrical strain gages cemented to it. The beam was made of 75ST-Aluminum. The strain gages used were SR-4 (AB-7) gages. By using two gages, one on either side of the beam, a reading of twice the actual strain was obtained. A Foxboro SR-4 strain time recording instrument was used to record the strain. Its power supply was furnished through a voltage regulator for more accurate results. Naturally, this type of apparatus must be calibrated to eventually give an equivalent friction torque reading. The component parts of the strain measuring device are shown in figures 6 and 7. Figures 11, 12, 13 and 14 are samples of the recording charts. The time for one complete revolution was thirty minutes.

The bearings which support the oil reservoir were mounted in a bulky cast aluminum base for the purpose of damping vibrations. See figures 2 and 3. This base was then mounted on a large ball-type thrust bearing in such a manner that the center of rotation of the spindle was located a certain distance from the center of rotation of the thrust bearing. In this way a load could be applied to the base that would create tension in the spindle driving belt. This force was applied by a cord attached to the base and passing over a pulley, the other end having various weights applied for varying the load on the spindle journal bearing. The cord is shown in figure 4. The place for applying the weights is shown in figures 4 and 5. The line of action of the applied load was in the same vertical plane as the action line of the load on the spindle and was parallel to the latter so that the load on the spindle was equal to the applied load, neglecting the friction in the pulley and thrust bearings. The friction was found to be negligible.

The bottom race of the thrust bearing could be moved parallel to the load line by means of a screw attachment shown in figures 4 and 5. In this way the direction of all loads could be maintained, during the run, as the driving belt stretched or contracted without changing the magnitudes of the loads. This is essential since a constant and accurately determined load is required for a bearing analysis.

Temperatures within the oil reservoir were measured by means of two thermocouples. One was silver soldered in a groove on the outside of the journal bearing, while the other was positioned loosely in the lower portion of the reservoir. The exact location of these thermocouples are shown in figure 2. A switching unit was used in conjunction with a Foxboro portable indicator to give direct temperature readings.

The ambient temperature was controlled by installing the equipment in a specially built air conditioning unit that controlled only the dry bulb temperature. Heating elements with a fine rheostat control were used since temperatures above 75 F were desired. A circulating fan was used to keep the temperature uniform throughout the enclosure. In this way temperatures could be regulated with negligible deviation.

Controlled speed of the spindle for such an investigation is also essential. This was accomplished by driving the belt with a synchronous motor through a specially built nine speed transmission. The output speeds of the transmission were from 2,000 to 6,000 revolutions per minute with an input speed of 1,800 revolutions per minute. The belt pulley ratio provided spindle speeds of approximately 4,000 to 12,000 revolutions per minute with increments of 1,000 revolutions per minute. The minimum allowable belt tension to avoid slippage was calculated to insure constant speeds.

In order to minimize vibration of the spindle an aluminum reservoir was built to replace the bobbin. The reservoir was to be filled with weighed amounts of mercury to provide various thrust loads on the conical thrust bearing. A screw fitting in the reservoir could be adjusted so that the mercury could not climb the walls of the partially filled reservoir.

To insure a smooth running belt, an idler pulley was mounted rigidly to run on the slack side of the belt.

All of the running equipment was mounted on a heavy cast-iron plate, which was designed for the purpose of eliminating the transfer of vibrational effects to the spindle. See figure 4. The plate has mounting holes for Newell's original test apparatus so that further data may be obtained on windage losses with similar environmental conditions.

Some of the apparatus as shown in figure 7 was designed to be used with Newell's apparatus. That part of the apparatus will not be discussed here.

CHAPTER III

EXPERIMENTAL PROCEDURE

Calibration of torque measuring device.—It was necessary to calibrate the torque measuring device because as the thread wraps around the base of the oil reservoir, with increasing frictional torque in the journal bearings, the direction of the cantilever beam changes slightly. Also to be considered is the fact that strain readings obtained with SR-4 strain gages on relatively thin sections cannot be accurately converted by strength of materials theory to give the applied load.

A small amount of static friction exists in the ball-bearings supporting the oil reservoir. If the torque measuring device is not calibrated correctly, this static friction will give incorrect results.

The first step in the calibration is to clean the ball-bearings thoroughly with a solvent. Then one or two drops of very light oil should be placed on the inner and outer races and the bearings rotated to distribute the oil. The purpose of the oil is to prevent oxidation of the metal. Any more oil tends to increase the static friction.

The next step is to wrap a thread around the base of the oil reservoir in a similar manner to the way the thread connecting the cantilever beam is wrapped, only in the opposite direction. This thread is then run over a ball-bearing pulley attached to the spindle mounting base. The pulley is shown in figure 3. Known weights are then applied to the thread to produce torques covering the expected range. In this way a calibration curve of torque versus strain can be obtained.

It is of utmost importance that the driving motor be running during the calibration. The resultant vibration is responsible for jarring loose the static friction in the ball-bearings. This condition would then be similar to the actual test condition.

Calibration of the ball thrust bearing.—If this bearing is kept clean as described above, there is no need for a calibration. The static friction is jarred loose by the slight amount of vibration in the table. As a result the load deviation due to static friction is negligible.

Speed determination.—Though the spindle is driven with a synchronous motor, it is necessary to determine the speed of the spindle with a stroboscope. This is because the belt pulleys do not give the exact speed ratios as indicated by the pulley diameters. As long as no slippage of the belt occurs the spindle speed will remain constant. Therefore, only one speed determination needs to be made for each output speed of the transmission. Frequent checks can be made during a run to verify the initial calibration and to check for belt slippage which might result from an oily belt or insufficient belt tension. During this investigation some belt slippage was allowed, but the speed remained constant.

Ambient temperature control.—The ambient dry bulb temperature is controlled by means of four heating elements, one of which is in series with an external rheostat. Each heating element is controlled by a separate switch.

Because of a considerable time lag in the temperature control, it is necessary to warm up the apparatus to the desired temperature some time before the test runs are made to attain thermal equilibrium. This must be done with the spindle running. As the power consumption of the spindle increases the output power of the heating elements must be decreased.

Oil temperature and viscosity determination.—The length of thread on the end of the cantilever beam should be adjusted so that, when the oil reservoir is in equilibrium with all the loads, the thermocouple on the journal bearing will be in the area of the maximum pressure region of the oil film. This will insure a maximum temperature reading, when the thermocouple leads are connected. The temperatures may be read only after thermal equilibrium has been reached, and only when torque readings are not desired. When the thermocouples are secured to the connecting post on the top of the spindle base, they interfere with the torque readings.

After calculating the approximate temperature of the oil film, the average viscosity of the journal bearing oil film can be read directly from the ASTM chart which was plotted from the data given for the various oils.

Test procedure.—The test procedure was as follows:

1. The oil reservoir was first filled with the most viscous oil to avoid the possibility of burning out the bearings at high speeds before most of the data could be obtained.

2. The synchronous motor, heating elements and circulating fan were turned on for the purpose of obtaining thermal equilibrium at the desired running temperature. During this time the thermocouples in the journal bearing and oil reservoir were connected so that thermal equilibrium could be checked in the spindle. Thermal equilibrium was checked in this manner before each run. In the case where the loads on the bearings were varied, the loads were added before attempting to establish thermal equilibrium.

3. When thermal equilibrium was attained, the thermocouple leads were disconnected from the connecting post in order to free the oil reservoir for strain readings. The strain was then recorded on the time recorder for approximately twenty minutes.

At the end of each run the thermocouples were again connected, the reservoir positioned to obtain the maximum temperature reading, and the reading recorded. This was for the purpose of rechecking the thermal equilibrium.

Types and order of runs.—Two series of data were desired for this investigation. They were obtained by:

1. Varying the speed with constant loads for five different oils;

2. Varying the load on the journal bearing using the

lowest speed and least viscous oil.

The data were obtained in the same order as listed above.

CHAPTER IV

DISCUSSION OF RESULTS

<u>Performance of equipment</u>.—One of the main purposes of this investigation was to develop the apparatus capable of accurately obtaining the necessary data. Though not much time was left for proving the capabilities and accuracy of the equipment, the little data that was obtained indicates that accurate and reproducible data can be obtained. Several calibrations of the torque measuring device show that the static friction in the bearings supporting the oil reservoir is negligible, if special care is taken to keep the parts clean. Also, to obtain best results excessive tension in the driving belt should be avoided so as not to increase the load on the bearings.

The results of the torque measuring device calibration curves look very good. Calibrations were made before and after the test runs. The results were the same each time. Figure 8 is the average calibration curve used for obtaining the data in this report. It is noticed that a straight line was obtained. All calibration curves for this device should appear as such, for it signifies that the stress in the aluminum beam is well below the yield stress. (The major portion of a stress strain diagram for aluminum alloys of the type used is a continuous smooth curve.) The importance of the a-

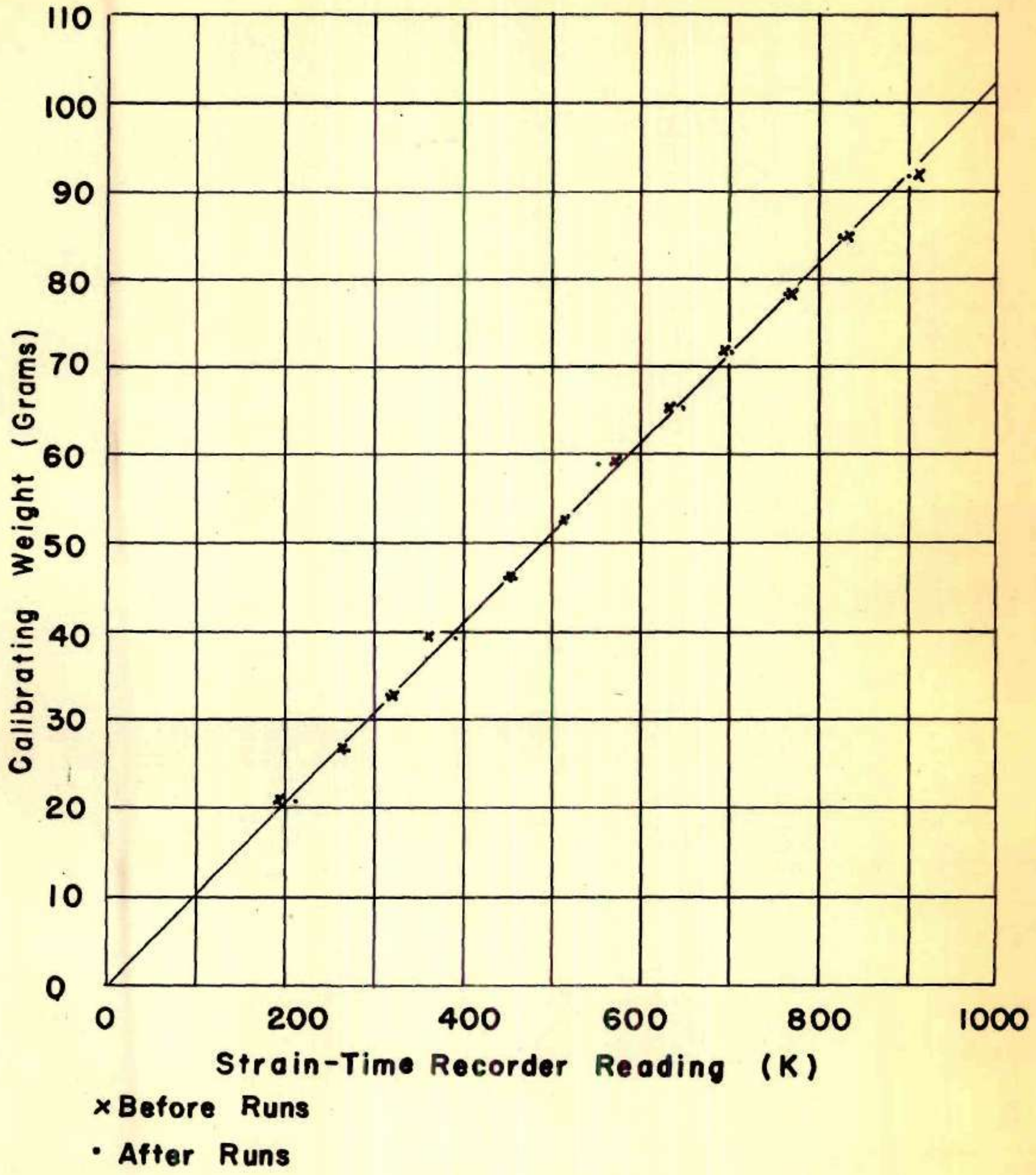


Fig. 8. Calibration Of Torque Measuring Device

bove condition lies in the fact that SR-4 strain gages tend to creep when subjected to fairly high strains. This is most likely not to occur when the stress strain diagram obtained is a straight line. The straight line also indicates that the deflection of the cantilever beam did not appreciably change the direction of the applied force. If this had occurred, the rate of change of torque with respect to strain in the beam would have changed in such a way as to make the instrument less sensitive at the higher values of friction torque.

Most of the above conditions can theoretically be calculated before making any runs. However, the test results indicate that the more accurate results are obtainable when frequent checks are made on the apparatus.

A certain amount of vibration was detected in the spindle. Preliminary runs were made with and without the mercury reservoir installed. No difference in vibration was detected. Since it was intended that several runs be made with mercury in the reservoir the reservoir was left on the spindle for all runs. It is now suspected, however, that the reservoir caused an excessive amount of vibration. Also, when an attempt was made to make runs with the reservoir partially filled with mercury, the mercury worked its way up into the empty compartment and caused considerable vibration. Undoubtedly, some other means for weighting the spindle will have to be devised.

Lubrication of the bearing.—The purpose of investigating the spindle was to establish the type of lubrication with which it generally operates. The curves in figure 9 indicate that the two bearings together operate as a hydrodynamic bearing. This is true because the curves rise to the right. As is shown in figure 1 hydrodynamic lubrication exists in the region where the coefficient of friction increases with an increase in the Sommerfeld number.

It should be noticed that the coordinates of this curve are not identically the same as for the general Sommerfeld curve as shown in figure 1. The value $\frac{r}{c}$ f has been replaced with K which is simply the time recorder chart reading. This can be done because the load and torque are constant. Therefore, K is directly proportional to $\frac{r}{c}$ f. The Sommerfeld number has been replaced with the spindle speed since the load is constant and for each oil the viscosity is nearly constant. If this had not been done, only one curve would appear instead of five. As shown, the curves also provide a relative comparison of power consumption.

The problem of determining the type of lubrication for each of the two bearings separately was easily solved by removing the spindle from its bearings and observing the pivot or pointed end. It was evident that metal to metal contact had been taking place, creating a condition of severe boundary lubrication. Only about one eighth of the area of the tip had made contact.

Curing the test runs it was noticed that the temperature of the oil bath near the step bearing was equal to and sometimes exceeded the temperature of the journal oil film.

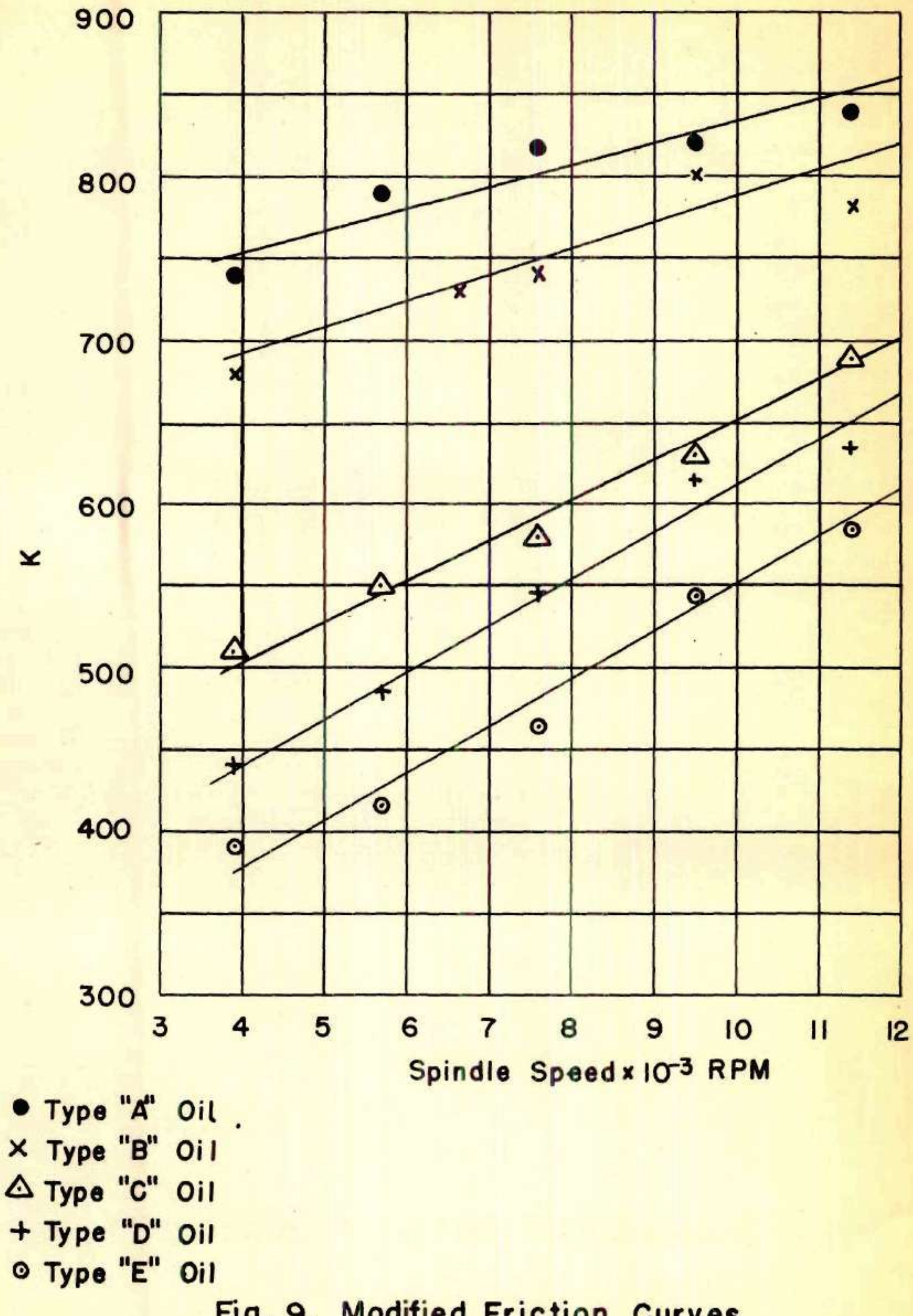
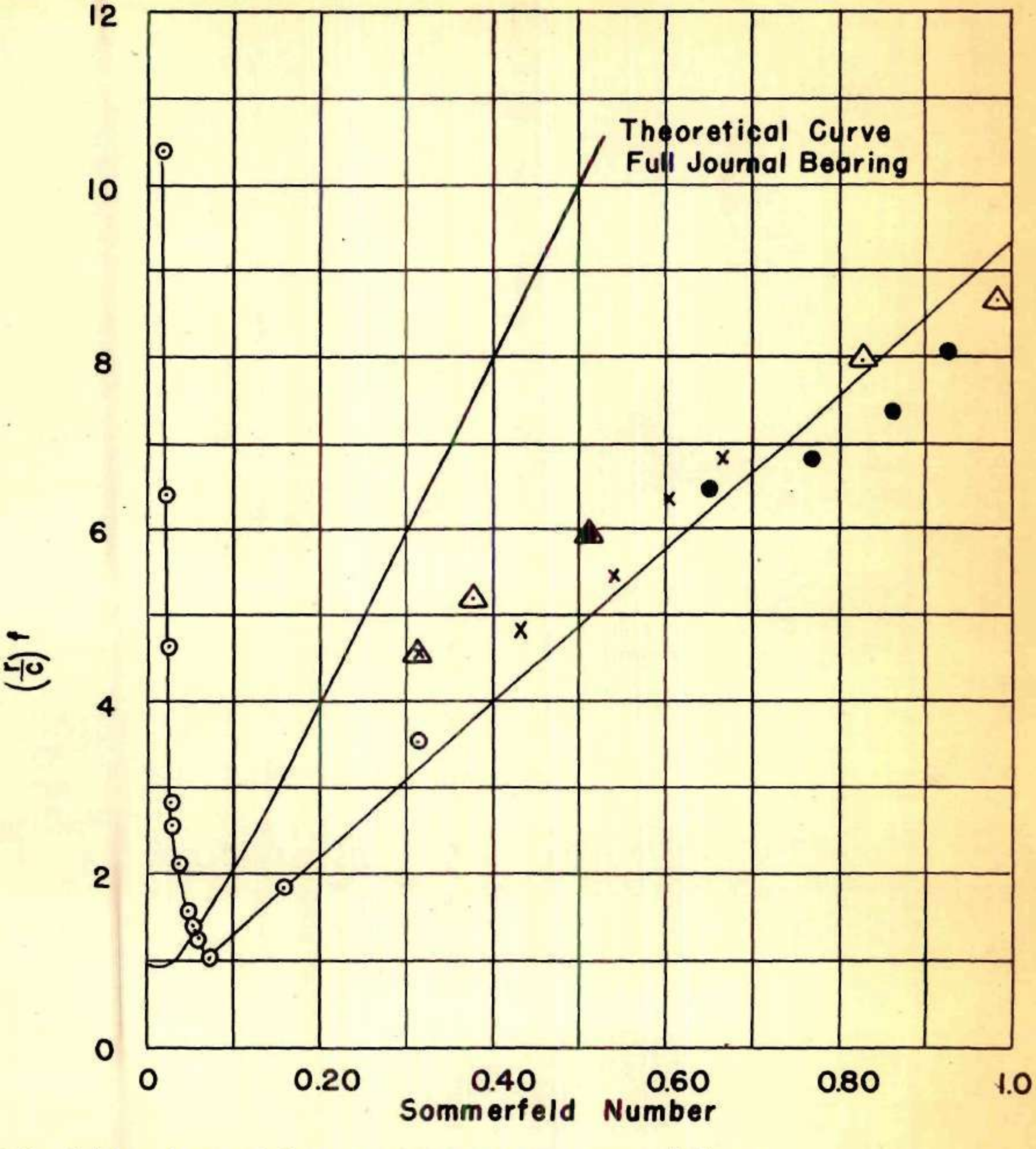


Fig. 9. Modified Friction Curves

Such a condition could exist only if the step bearing was operating with a high coefficient of friction.

Since the spindle used in this investigation was a new spindle it is possible that with sufficient running-in time the step bearing surface will wear down until metal to metal contact ceases. Actually, a step bearing should operate hydrodynamically[7].

Assuming that the journal bearing consumed the greater portion of the power, the friction curves in figure 10 were plotted. The theoretical curve was obtained by substitution of general values in the theoretical equations for a full journal bearing. The experimental curve was obtained as indicated on the graph. The curve is a very good illustration of the type of lubrication attainable with a spindle. However, it is not a true representation, even if the step bearing may be neglected, because, when converting from bearing to journal friction, values of n corresponding to a full journal bearing were used. It is evident that the experimental values of the coefficient are less than the theoretical in the hydrodynamic region. This is explained by the fact that partial bearings operate with a lower coefficient of friction in the hydrodynamic region. Also a partial bearing operates with a larger attitude, n, than a full journal bearing. Using the larger value of n to obtain the friction on the journal, knowing the friction on the bearing, larger values of f are obtained. This affects the boundary region more than the hydrodynamic since the n values for journal bearings decrease in



⊙ Variable Journal Load, RPM=3940, Type "E" Oil

× Variable Speed, Journal Load= 1.1 Lbs., Type "E" Oil

● Variable Speed, Journal Load= 1.1 Lbs., Type "C" Oil

△ Variable Viscosity, RPM=3940, Journal Load=1.1 Lbs.

Fig. 10. Spindle Friction Curves

magnitude with an increase in the Sommerfeld number. By trial and error a better fitting curve might be obtained. The critical or transition point would appear at the same Sommerfeld number, but the minimum value of the coefficient of friction would be greater.

The location of the critical point is very important because it is an unstable region in which to operate. A small decrease in the Sommerfeld number could result in a very high coëfficient of friction, which would mean excessive power losses and spindle wear. To obtain the critical point and boundary condition during the test, the load on the journal bearing was increased while running at the minimum speed with the least viscous oil. The critical point occurred with a journal load of about six pounds. The Sommerfeld number was approximately 0.07 as shown in figure 10. It is not likely that such a small Sommerfeld number would be encountered in practice because higher speeds are generally used in a spinning room. However, it should be remembered that three variables besides the dimensions of the bearings affect the Sommerfeld number. The same relative change in any one of them will give approximately the same change in the coefficient of friction. This is demonstrated in the hydrodynamic region of the curve by the plot of several points obtained by varying the different quantities u, N and P. According to theory, exact correlation should exist.

Vibrational effects.—One of the advantages of the torque measuring device is that it can record continuously the reaction of the spindle in relation to the oil film. Figures 11, 12, 13 and 14 are photographs of some actual records obtained during the test runs.

One interesting aspect of the device is that it appears to be capable of indirectly measuring relative magnitudes of vibration of the spindle. As yet this has not been proven, but it is interesting to examine figures 11 and 12. Figure 11 is a record of the spindle running with the mercury reservoir partially filled. During the run, centrifical forces caused the mercury to leak into the upper chamber of the reservoir, causing vibration of the spindle. The amplitude naturally increased with an increasing amount of eccentrically displaced mercury. The figure shows this trend by means of an increasingly widened inked line. The recording pen was vibrating with a very low frequency, which could have been a harmonic of the spindle frequency.

Figure 12 is a complete record of one oil with variable spindle speed. Though each recording was for only a six minute duration, a sufficient length of time was allotted between runs for thermal equilibrium to be obtained. It is observed that the second torque reading is not equally spaced radially as the others. Besides giving too large a reading, the inked line is thicker than the rest. It is reasonable to suspect then that vibration of the spindle is causing the abnormally high friction. This is also probably the cause of the rough distribution of points in figures 9 and 10. Figures 13 and 14 are very good illustrations of the effects of vibration on

Figure 11

Strain-Time Recording Chart

Amplitude of Vibration Increasing

Mercury Reservoir Partially Filled

Spindle Speed = 11,400 RPM

Type Oil: A

Journal Load = 1.1 lbs.

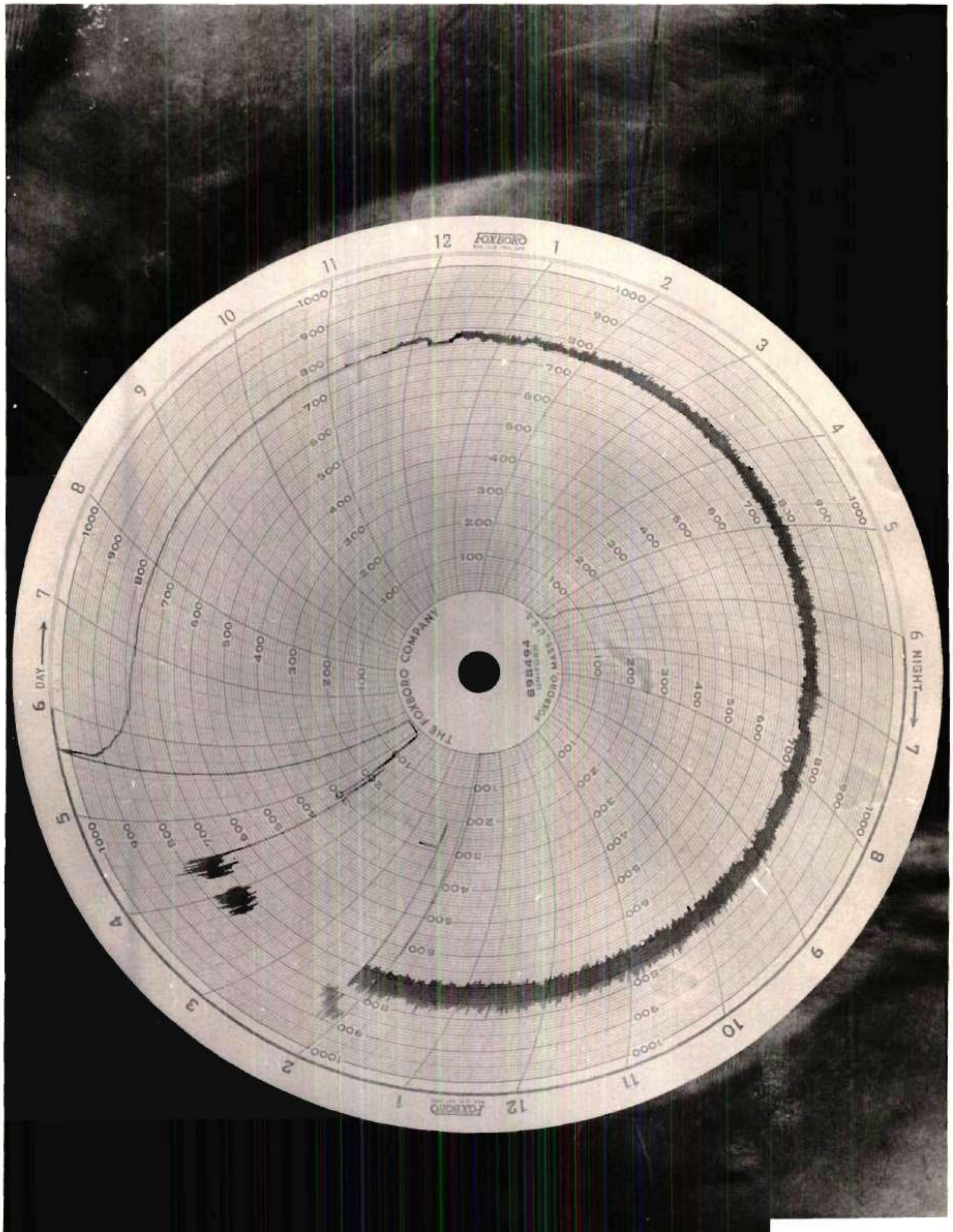


Fig. 11

Figure 12

Strain-Time Recording Chart

Increasing Friction Torque

Empty Mercury Reservoir

Spindle Speeds = 3940, 5700, 7600, 9500, 11,400 RPM

Type Oil: E

Journal Load = 1.1 lbs.

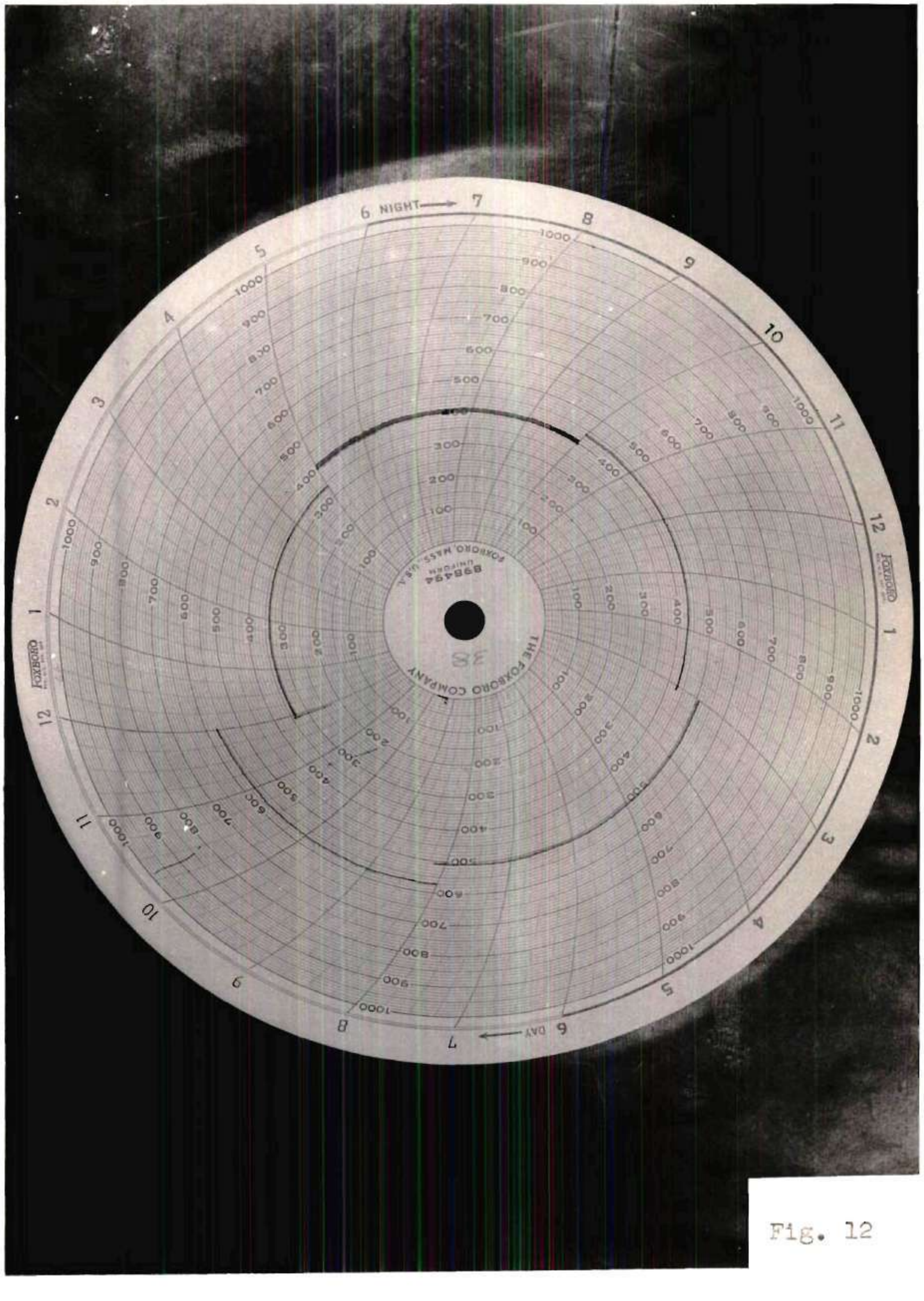


Fig. 12

Figure 13

Strain-Time Recording Chart

Vibration and Damping

Empty Mercury Reservoir

Spindle Speed = 5700

Type Oil: A

Journal Load = 1.1 lbs.

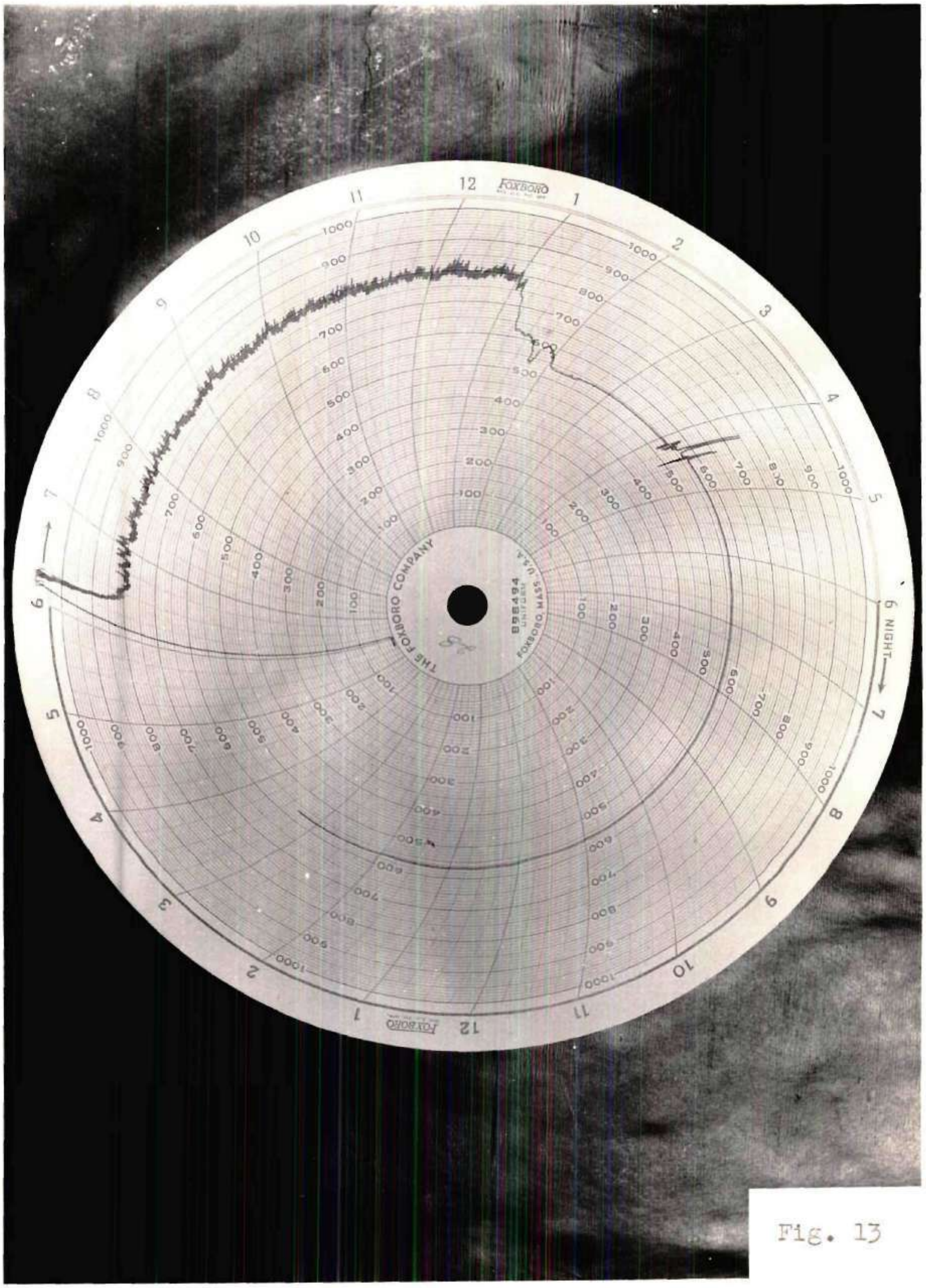


Fig. 13

Figure 14

Strain-Time Recording Chart

Irregular Vibration and Damping

Empty Mercury Reservoir

Spindle Speed = 7600 RPM

Type Oil: A

Journal Load = 1.1 lbs.

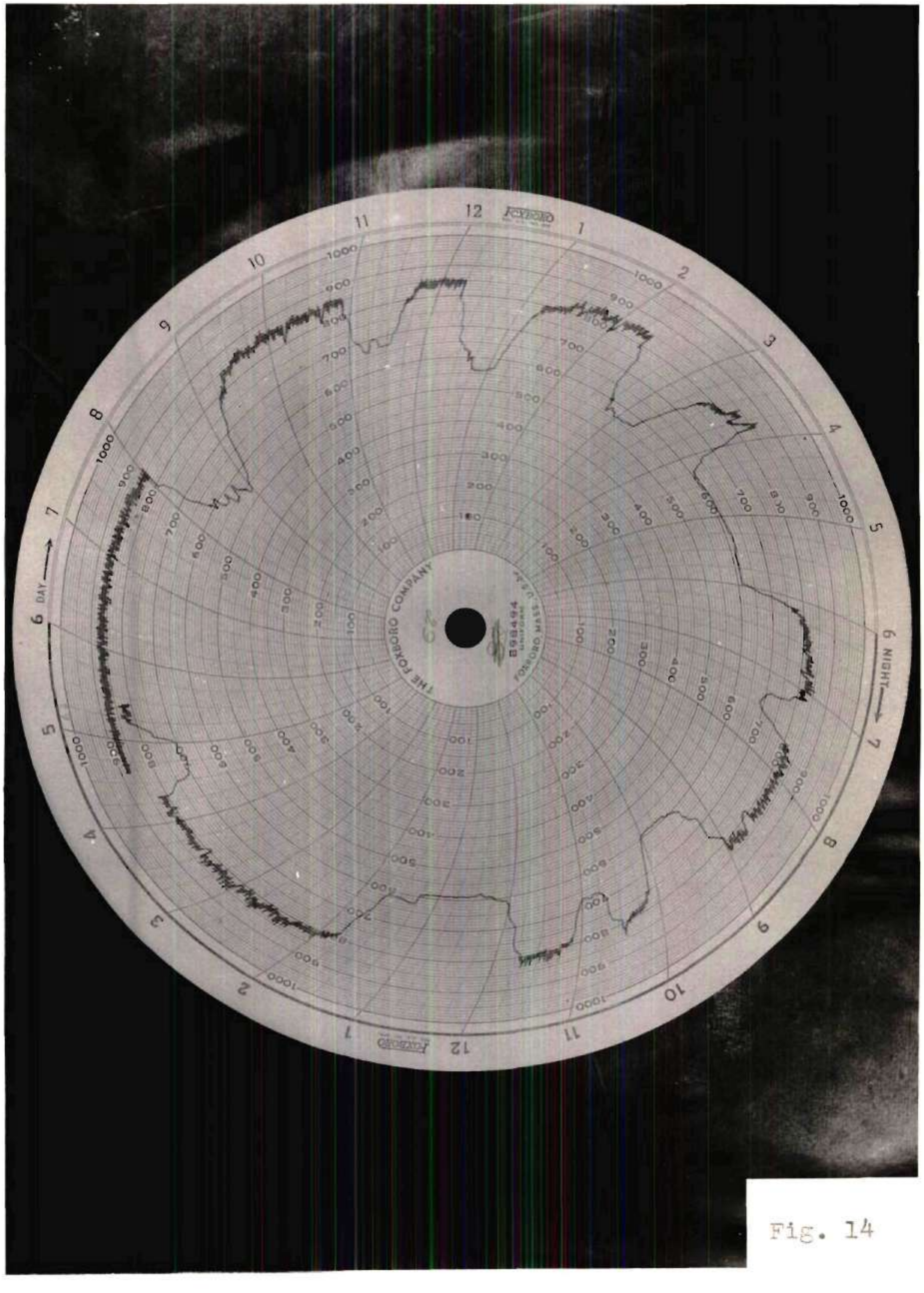


Fig. 14

the power consumption. These were obtained using the most viscous oil. There was a difference of approximately 2,000 rpm in their speeds. Of course it is apparent that damping took place. The irregular damping in figure 14 indicates that conditions were such that a critical damping region existed. A condition such as this could certainly be detrimental to the production of uniform thread.

After several tests, it was found that the critical damping as shown in figures 13 and 14 occurred only for the two particular speeds and one oil. Since the oil used was the most viscous of the five, it is possible that complete damping took place only for the two runs.

CHAPTER V

CONCLUSIONS

The following conclusions are based wholely on the results of the investigation:

1. The apparatus developed for this investigation is satisfactory for accurately measuring the actual magnitudes of the variables involved rather than just relative magnitudes.

2. The plain cast-iron bolster bearings as a unit operate as a hydrodynamic bearing.

3. The plain journal bearing operates as a partial hydrodynamic bearing.

4. The step bearing, during this investigation, operated in the extreme boundary lubrication region.

5. A possible redesign of the step bearing to provide hydrodynamic lubrication is warranted.

6. Spindles in use today should be checked to determine how close to the critical point on the friction curve they are operating, a minimum value of 0.07 being considered unstable.

CHAPTER VI

RECOMMENDATIONS

Since this investigation did not include the effects of all the normal operating loads on the spindle it is recommended that the work be extended to include the effects of controlled amounts of vibration, thread tension load, and step bearing load. These added variables may narrow down the possible range of hydrodynamic lubrication.

After a much extended running-in period for the step bearing, it is recommended that this bearing be thoroughly analyzed by changing the vertical loading to determine if it will eventually operate hydrodynamically. If it will not, a change in design should be attempted. Also some work should be done with boundary lubricants in an effort to reduce friction in the step bearing as long as it operates in the boundary region.

If the above recommendations can eventually be carried out on the same spindle and plain cast-iron bolster, a complete and basic analysis will have been made. This information should then reveal the type of lubricant necessary.

In addition to the above recommendations it is suggested that some thought be given to the theory of oil whip[9] which is a self excited vibration found in lubricated journal type bearings. It could very well be that such a condition is partly responsible for the vibration of the spindle.

APPENDIX

Table 1. Tabulated Data For Modified Friction Curves

Constant Load on Journal Bearing = 1.1 Lbs.

Constant Load on Step Bearing = 0.855 Lbs.

RPM Spindle	TEMPERATURES Journal °F	Oil Bath °F	Ambient °F	K
		TYPE OIL: A		
3940	95	95	85	740
5720	99	98	85	790
7610	105	105	85	820
9500	110	110	85	820
11,400	115	115	85	840
		TYPE OIL: B		
3940	95	95	85	680
6650	105	105	85	730
7610	108	106	85	740
9500	110	111	85	800
11,400	111	114	85	780
		TYPE OIL: C		
3940	90	88	85	510
5720	99	96	85	550
7610	100	100	85	580
9500	106	106	85	630
11,400	110	112	85	690
		TYPE OIL: D		
3940	93	93	85	440
5720	95	95	85	485
7610	99	100	85	545
9500	104	105	85	615
11,400	108	110	85	635
		TYPE OIL: E		
3940	90	90	85	390
5720	93	94	85	415
7610	96	96	85	465
9500	100	102	85	545
11,400	106	108	85	585

Table 2. Tabulated Data For Friction Curves

RPM Spindle	Type Oil	Jour. Load Lbs.	Jour. Temp. °F	S	f_b	$\frac{F_j}{F_b}$	f_j
3940	E	14.0	104	.018	.0320	5.70	10.40
3940	E	12.8	102	.021	.0290	3.88	6.42
3940	E	11.5	101	.024	.0258	3.17	4.67
3940	E	10.2	100	.027	.0190	2.62	2.84
3940	E	9.3	99	.029	.0189	2.35	2.54
3940	E	8.2	98	.036	.0190	1.95	2.12
3940	E	6.6	97	.047	.0187	1.49	1.59
3940	E	6.2	96	.051	.0174	1.41	1.40
3940	E	5.7	95	.054	.0182	1.33	1.38
3940	E	5.3	94	.057	.0185	1.30	1.37
3940	E	4.6	93	.071	.0151	1.19	1.03
3940	E	2.2	90	.158	.0314	1.03	1.85
3940	E	1.1	90	.314	.0623	1.00	3.55
3940	E	1.1	90	.316	.0800	1.00	4.56
5700	E	1.1	93	.432	.0854	1.00	4.87
7600	E	1.1	96	.542	.0954	1.00	5.44
9500	E	1.1	100	.604	.1114	1.00	6.36
11,400	E	1.1	106	.662	.1200	1.00	6.85
3940	C	1.1	90	.511	.1045	1.00	5.97
5700	C	1.1	99	.652	.1130	1.00	6.45
7600	C	1.1	100	.771	.1190	1.00	6.80
9500	C	1.1	106	.864	.1290	1.00	7.39
11,400	C	1.1	110	.928	.1520	1.00	8.66
3940	E	1.1	90	.314	.0800	1.00	4.56
3940	D	1.1	93	.378	.0909	1.00	5.19
3940	C	1.1	90	.512	.1045	1.00	5.97
3940	B	1.1	95	.830	.1400	1.00	8.00
3940	A	1.1	95	.985	.1520	1.00	8.66

Table 3. Calibration of Torque Measuring Device

Wt. No.	Individ. Wt. gms.	Sum of Wts. gms.	Strain-Time Recorder Reading (K) Before runs	After Runs
1	20.688	20.688	190	215
2	6.103	26.791	265	270
3	6.055	32.846	320	315
4	6.750	39.596	360	390
5	6.693	46.289	450	455
6	6.192	52.481	510	515
7	6.652	59.133	570	550
8	6.173	65.306	630	645
9	6.594	71.900	690	700
10	6.651	78.551	770	760
11	6.502	85.053	830	825
12	6.680	91.733	910	900

Table 4. Spindle Oils

(Courtesy of The Texas Company)

Code Letter	E TL-1512	C TL-1513	A TL-1514	D TL-1515	B TL-1517
Code No.-PAL	2884-L-52	2885-L-52	2886-L-52	2812-L-52	2813-L-52
Grav.	30.4	29.1	29.0	32.1	30.5
Flash	355	370	415	360	430
Fire	390	410	460	395	475
Visc., Kin.					
at 100 °F	13.32	21.28	46.38	17.13	39.54
at 130 °F	8.40	11.63	22.67	9.76	19.70
at 210 °F	2.96	3.87	6.27	3.49	5.70
Visc., Saybolt Universal Seconds					
at 100 °F	70.8	102.9	215	85.6	179
at 130 °F	53.5	64.7	109	58.1	96.4
at 210 °F	36.1	39.0	46.7	37.8	44.9
Color 6" Lov.	40	55	80	10	10
Pour	10	20	10	25	-5
Ash	None	None	.001	None	None

Technical Information

Test Spindle:

Saco-Lowell McMullan Spindle equipped with (1) plain cast-iron bolster with cast-iron steps and guides, or (2) plain steel tubing bolster with porous-metal inserts for steps and guides.

Electrical Strain Gages:

SR-4 (AB-7) gages

Gage factor = 2.00 ± 1 per cent

Resistance = 120 ± 0.5 ohms

Lot number 236

Spindle Oils:

See Table 4.

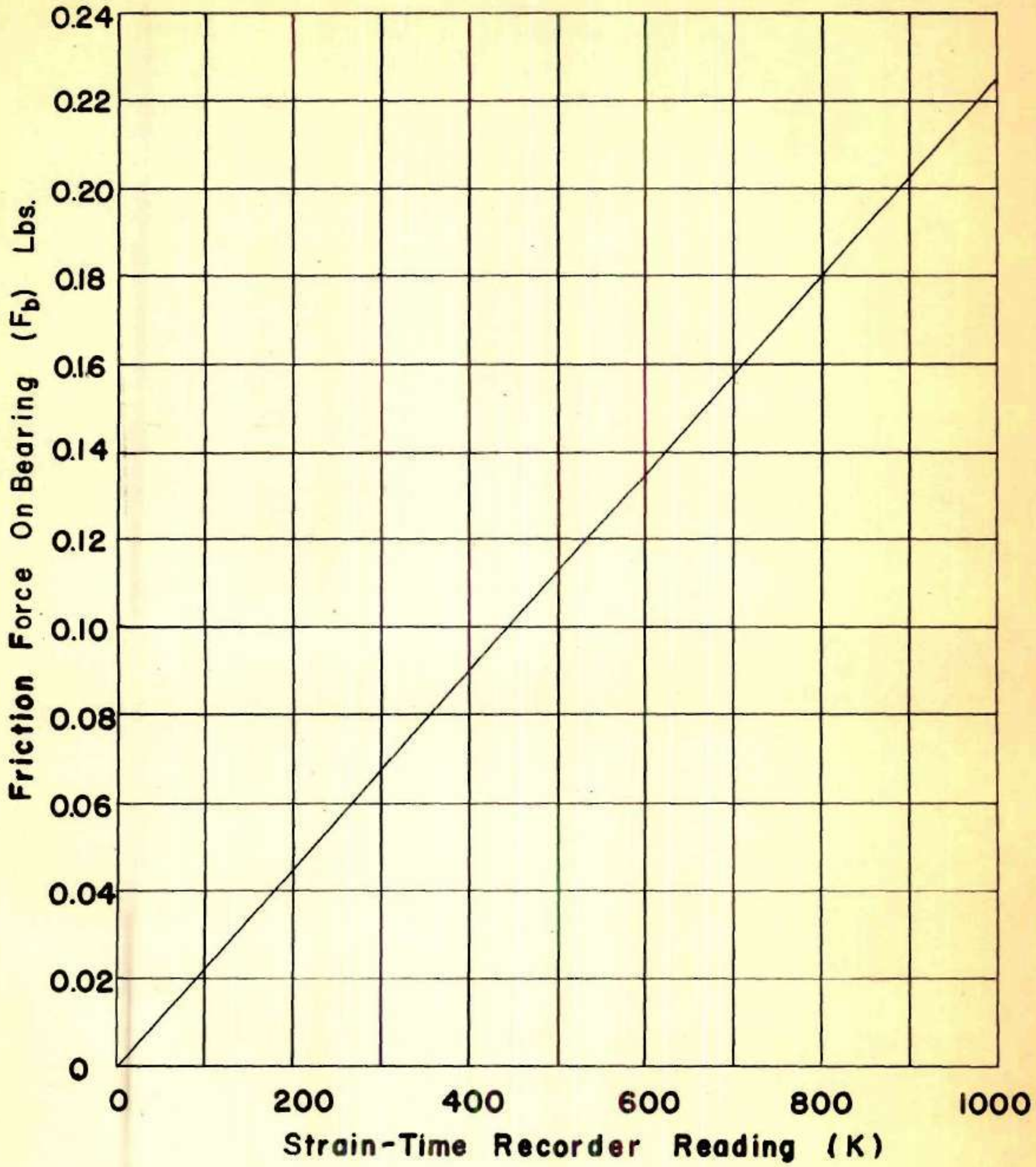


Fig. 15. Conversion Curve $K - F_b$

Sample Calculations

1. Absolute viscosity determination in the journal bearing

It is assumed that all the friction power consumed is dissipated as heat through the journal bearing.

For the case of maximum power consumption:

Power = Friction torque x RPM

$$= .433 \text{ in.} \times .19 \text{ lbs.} \times 11,400\frac{\text{rev.}}{\text{min.}} \times \frac{2\pi}{\text{rev.}} \times \frac{\text{ft.}}{12\text{in.}} \times \frac{\text{Btu}}{778\text{ft.lbs.}} \times \frac{60\text{min.}}{\text{hr.}}$$

$$= 37.9\frac{\text{Btu}}{\text{hr.}}$$

The change in temperature across the bearing rim is determined as follows:

$$t = \frac{q}{2\pi KL} \ln \frac{r_o}{r_i}$$

where q = total heat transferred = $37.9 \frac{\text{Btu}}{\text{hr.}}$

K = thermal conductivity of cast-iron = $35 \frac{\text{Btu}}{\text{hr.ft.}^\circ\text{F}}$

L = length of bearing = 2.13 inches

r_o = outside radius of rim (to thermocouple)

= 0.240 inches

r_i = inside radius of bearing = 0.180 inches

$$t = \frac{37.9 \text{ Btu/hr.} \quad 12 \text{ in./ft.}}{2\pi \times 35 \text{ Btu/hr. ft. }^\circ\text{F} \times 2.13 \text{ in.}} \ln\frac{0.240}{0.180}$$

$$= .28\,^\circ\text{F}$$

The change in temperature is negligible. Therefore, the indicator temperature reading for the journal bearing can be used for the journal oil film temperature.

2. Calculating the Sommerfeld Number for a full journal bearing.

$$S = \left(\frac{r}{c}\right)^2 \frac{uN'}{P'}$$

where r = radius of journal = 0.180 in.

c = radial clearance

$= \frac{.3650-.3587}{2} = 0.00315$ in.

u = absolute viscosity of oil, $\frac{\text{lb. sec.}}{\text{in.}^2}$

N' = RPS

P' = load per unit projected bearing area, lbs./in^2.

$$S = \left(\frac{0.180}{0.00315}\right)^2 \cdot \frac{\frac{uN}{P}}{.360 \times 2.13} \times \frac{1}{60} = 41.7 \frac{uN}{P}$$

where N = RPM

P = load on journal, lbs.

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